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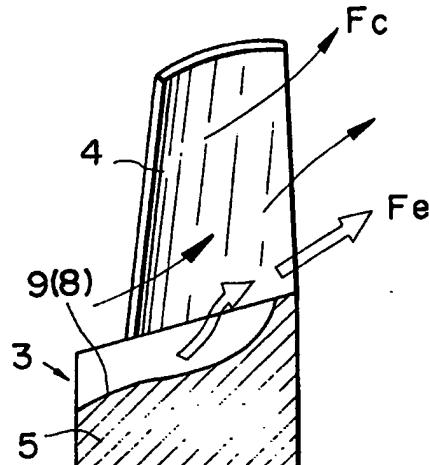
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(54) A radiator cooling fan

(57) A radiator cooling fan for an automotive vehicle comprises a hub portion 5 rotatably provided at the center of the fan, and a plurality of circumferentially spaced fan blades 4 extending radially outwardly from said hub portion, the pitch angle of said fan blades being so large (preferably 60–70°) at the base portion thereof, as to produce substantially a cone-line air current, the imaginary vertex of which is positioned upstream of the cooling fan, and the pitch angle thereof becoming gradually smaller from said base portion for said fan blades to the tip thereof. Spiral grooves 9 may be provided on the hub 5 to reinforce the air flow in this vicinity. Instead, the hub 5 may be provided with small auxiliary blades.

FIG. 11



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FIG.1

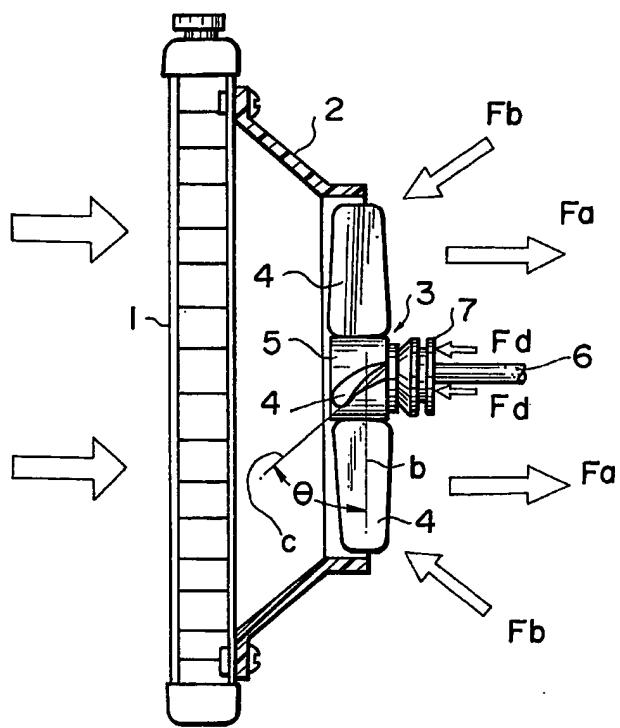
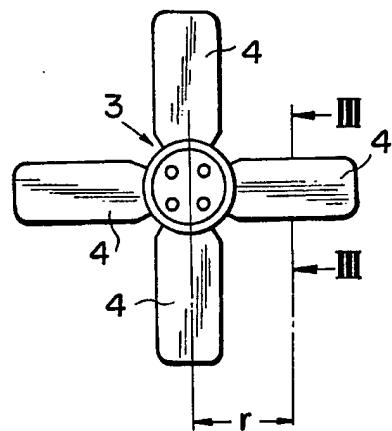


FIG.2



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FIG. 3

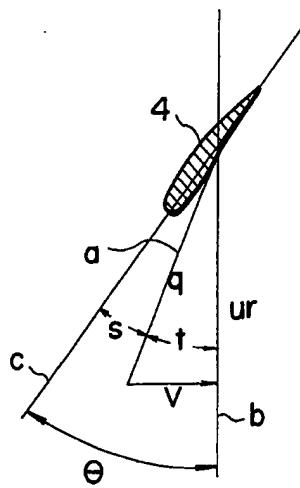


FIG. 4

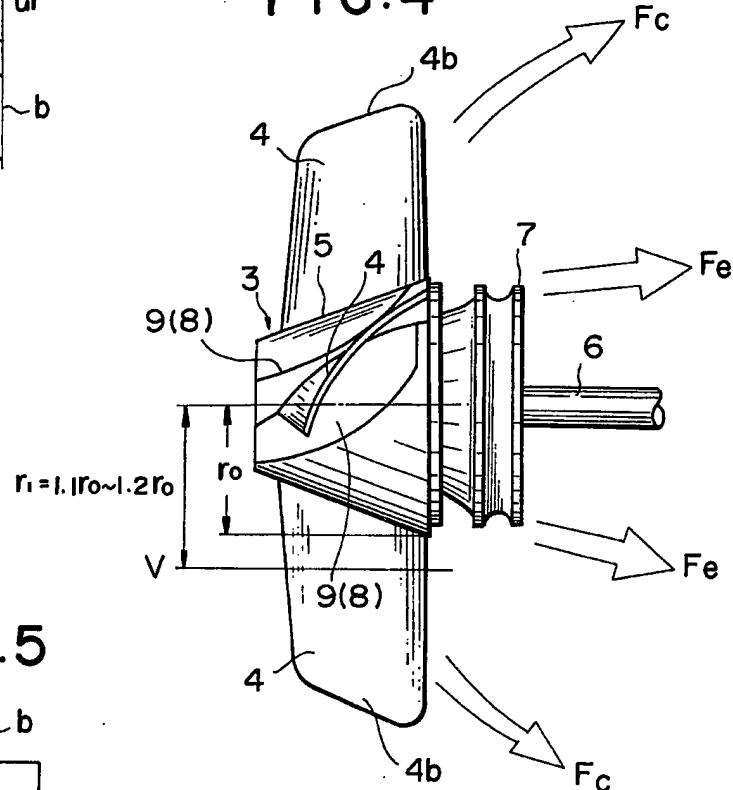
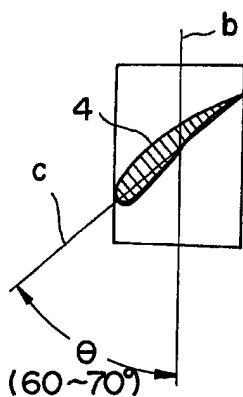


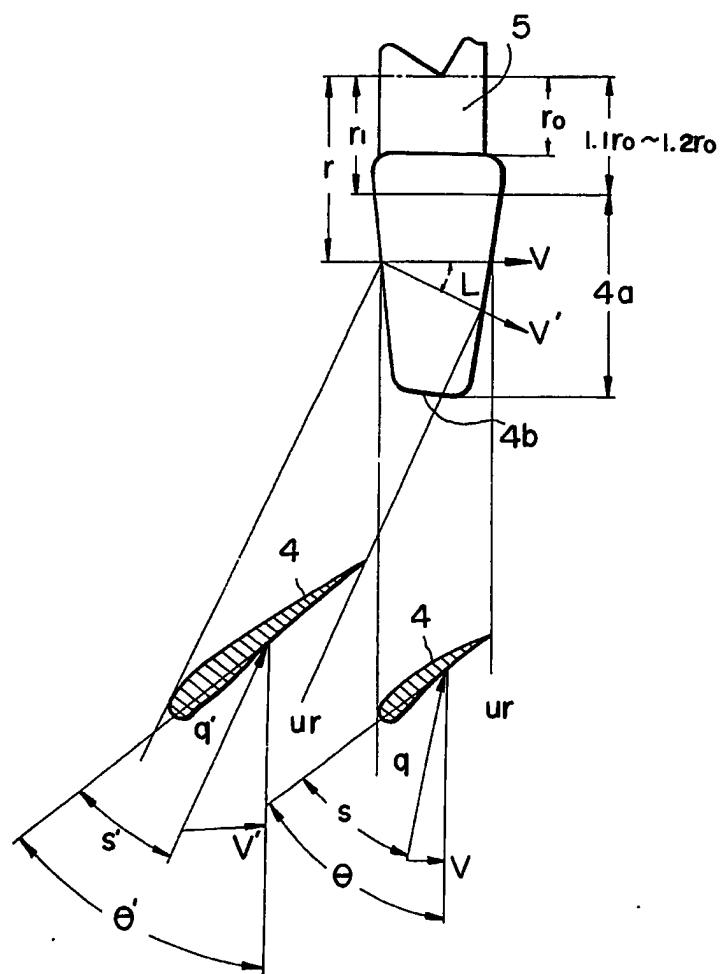
FIG. 5



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FIG. 6



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FIG. 7

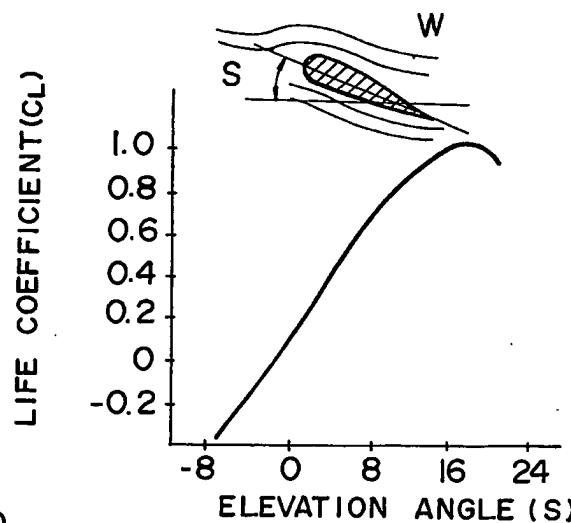


FIG. 8

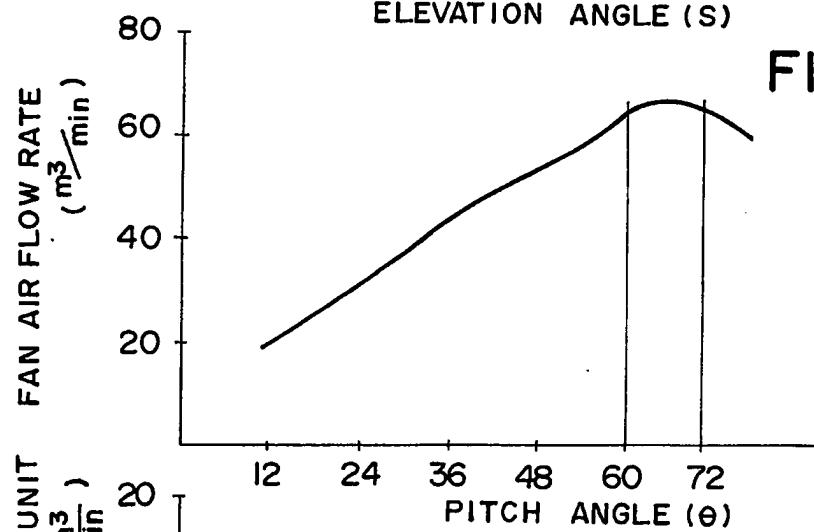
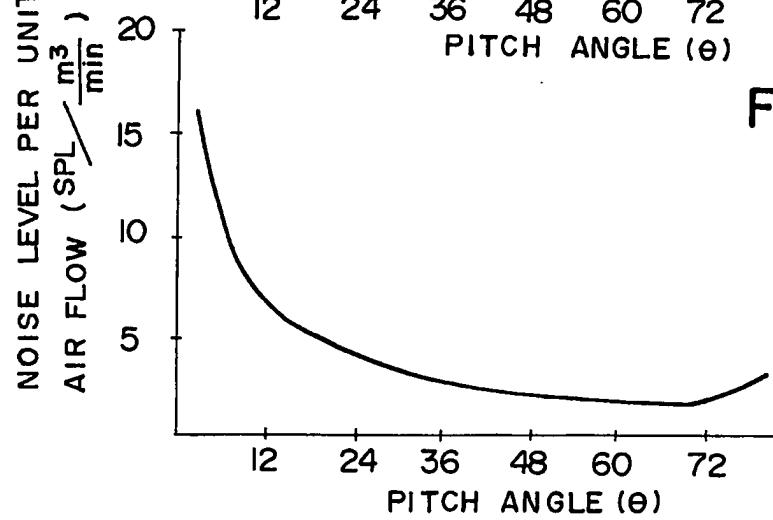


FIG. 9



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FIG. 10

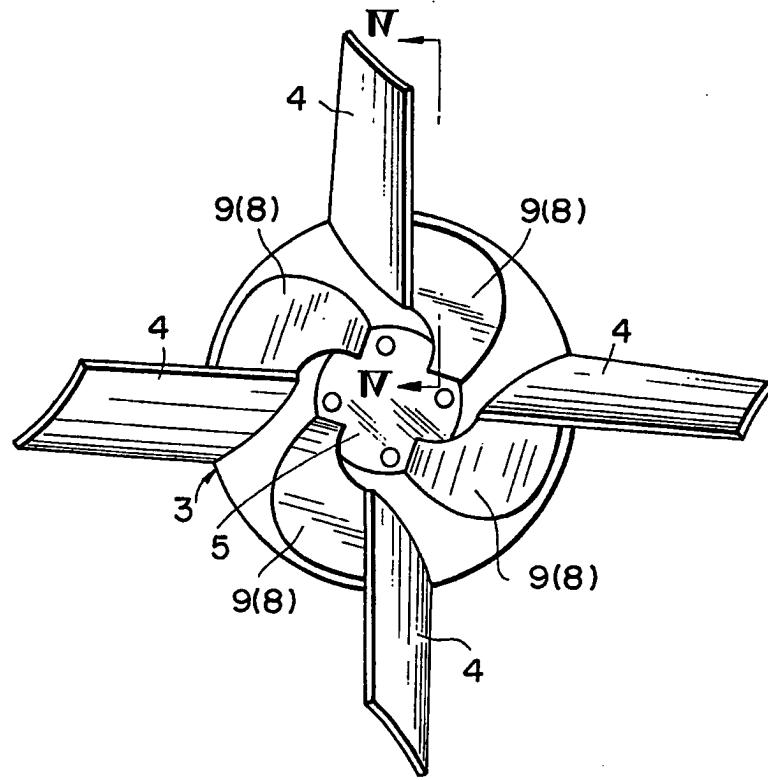
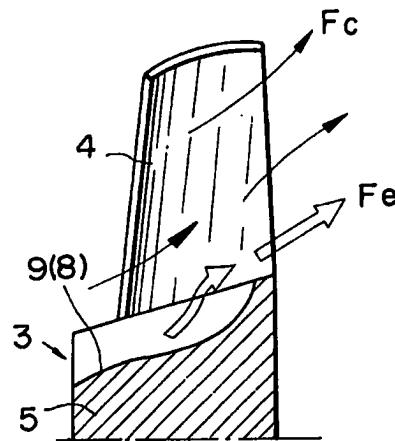


FIG. 11



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FIG.12

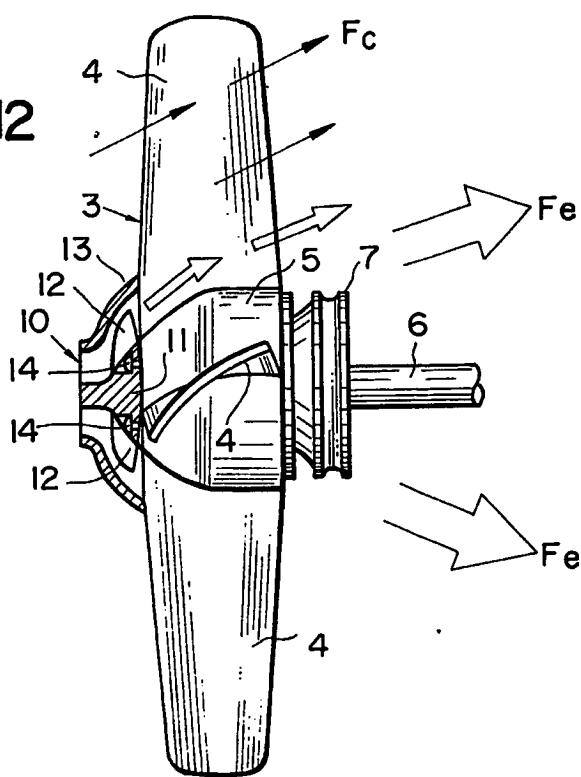
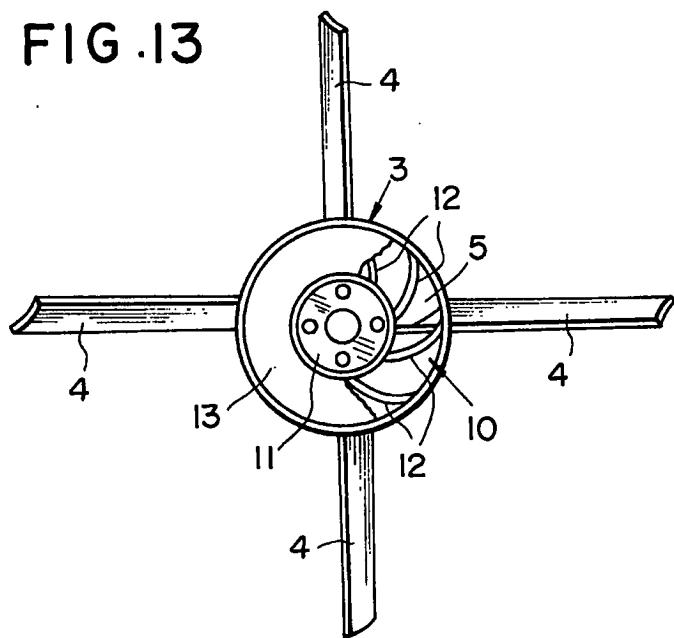


FIG.13



SPECIFICATION

A radiator cooling fan

5 The present invention relates to a radiator cooling fan for an automotive vehicle.
 In general, as a radiator cooling fan for an automotive vehicle, an axial flow type cooling fan equipped with a plurality of fan blades has
 10 been often used.

Fig. 1 shows a conventional construction of such a cooling fan. The radiator 1 is conventional and the fan shroud 2 is fixed at the rear of the radiator 1. The cooling fan 3 is provided in the outlet of the fan shroud 2 and has a fan coupling 7 which is driven through a crank pulley and a belt (not shown). A shaft 6 is rotatably arranged in a conventional manner. Also, the cooling fan 3 is composed of a boss portion 5 at the center thereof and a plurality of circumferentially spaced fan blades 4 which extend radially outwardly from the boss portion 5.

In the prior art axial flow type cooling fan, 25 the pitch angle θ between the chord c of the fan blades 4 and the rotation plane b thereof is set between 20° and 35° so that the air can flow in parallel to the fan shaft 6 as shown by the arrow F_a in Fig. 1.

30 Fig. 2 is a front view of the cooling fan 3 and Fig. 3 is a sectional view of the cooling fan taken along the line III-III in Fig. 2. r is the distance between the center of the cooling fan 3 and the sectional plane of the fan blade 35 4.

When the rotational angular velocity of the fan is U , the circumferential velocity at the sectional plane of the fan blade 4 is U_r . If the air flows at the velocity v from upstream of 40 the cooling fan to downstream thereof by the rotation of the fan blades 4, it becomes a composite air current as shown by the line a in Fig. 3 to run against the surface of the fan blades 4. Therefore, the velocity q of the 45 composite air current can be shown by the following equation:

$$q = \sqrt{V^2 + U^2 r^2}$$

50 The relationships of the direction a of the composite air flow, the rotation plane b of the fan blades 4 and the chord c can be shown by the following equation:

$$55 \quad \tan t = \frac{V}{U_r}$$

60 where t is the flow angle of the composite air current between the flow direction a of the composite air current and the rotation plane b of the fan blades 4.

If s is the elevation angle between the direction a of the composite air flow and the 65 chord c , θ the pitch angle between the chord

c and the rotation plane b , and n the rotational speed of the fan blades, then the rotational angular velocity U of the fan is $2\pi n$ so that the elevation angle s can be shown by

70 the following equation:

$$s = \theta - t = \theta - \tan^{-1} \frac{V}{2\pi n r}$$

75

This equation shows the fact that the characteristics of the cooling fan 3 are determined by the pitch angle θ and the value of $v/2\pi n r$. When the velocity v of the air flow from

80 upstream of the cooling fan 3 is constant, the smaller the distance r , the larger the value of $\tan^{-1} v/2\pi n r$. In other words, the value of $\tan^{-1} v/2\pi n r$ increases as it is close to the base of the fan blades.

85 Fig. 7 shows the relationship between the elevation angle s and the lift coefficient C_L when a normal blade W is positioned in the air flow. As can be seen from the graph in Fig. 7, the lift coefficient C_L is maximum when the 90 elevation angle s of the blade W is set between 15° and 20° . When the elevation angle s is more than such a value ($15-20^\circ$), the air flow rate decreases because of the air current separation phenomenon so that the efficiency 95 of the blade becomes low.

In the prior art axial flow type cooling fan, therefore, the pitch angle θ is set in such a way that the elevation angle s may be maintained constant at each sectional plane of the

100 fan blades 4 whereby the air can flow in parallel to the fan shaft so as to improve the efficiency of the fan blades without any separation phenomenon of the air current. In such an axial flow type cooling fan, in order to

105 maintain the elevation angle s constant at each sectional plane of the fan blades 4, the pitch angle of the fan blades 4 is set large at the base portion thereof and reduces toward the head of the fan blades.

110 However, if the pitch angle is set in such a way, the air flows in parallel to the fan shaft and runs directly against the engine and others arranged downstream of the cooling fan. That is, the engine and others are obstacles to

115 the air flow so that the air cannot flow smoothly. Thus, the air flow resistance increases and the required efficiency of the fan blades cannot be obtained in such a case. Also, the fan noise is relatively large because

120 not only the air flows in parallel to the fan shaft to run directly against the engine and others to thereby become turbulent therearound but also the reverse air flow occurs near the head of the fan blades 4 as shown by the 125 arrow F_b in Fig. 1. It is one of the largest noises occurring from the vehicle.

In order to avoid such fan noise, the diameter of the fan must be so enlarged that the air current flowing in parallel to the fan shaft 130 does not interfere with the engine and others

located at the rear of the cooling fan to some extent. Also, the rotational speed of the fan blades is limited to a reasonable degree. In such a case, however, the weight of the 5 cooling fan and the production cost increase. In addition, the cooling fan cannot be freely enlarged because of the restricted space for the attachment of the radiator and others.

Therefore, it is an object of the present 10 invention to provide a radiator cooling fan for an automotive vehicle in which the foregoing defects of the prior art can be overcome.

Another object of the present invention to 15 provide a cooling fan in which the air can flow smoothly from upstream of the cooling fan to downstream thereof by minimizing the air flow resistance to an engine of an automotive vehicle and others.

A further object of the present invention is 20 to provide a radiator cooling fan for an automotive vehicle in which the fan noise can be prevented.

Still a further object of the present invention 25 is to provide a cooling fan for an automotive vehicle in which the efficiency of the cooling fan can be increased.

An additional object of the present invention 30 is to provide a cooling fan for an automotive vehicle in which the mass rate of air flow by the cooling fan can be increased.

According to the present invention, a cooling 35 fan for an automotive vehicle includes a rotatable boss portion and a plurality of fan blades extending radially outwardly from the boss portion. The boss portion is positioned at the center of the cooling fan and the fan blades are circumferentially spaced on the boss portion. The pitch angle of the fan blade is predetermined in a special way. That is, the 40 pitch angle of the fan blade is relatively large (60 to 70° for the best results) at the base portion thereof which is effective as a rotation blade but becomes gradually smaller from the base portion to the head thereof. Thus, the air flow caused by the cooling fan becomes a cone-like air current, the imaginary fine point of which is positioned upstream of the cooling fan. In other words, the air flows obliquely or 45 divergently backwards from the cooling fan so that the air resistance to the engine arranged downstream of the fan decreases. This results in the improvement of the fan noise and performance. For the best results, the cone of the air current has a taper as shown in the 50 Figures.

It is preferable that air flow guide means is 55 formed on the fan boss portion in order to prevent the air pressure from dropping down at the back of the cooling fan as the result of the air flow becoming divergent.

These and other objects, features and advantages of the present invention will become 60 more apparent from the following description of the preferred embodiments thereof when read in conjunction with the accompanying

drawings in which:

Figure 1 is a schematic sectional view showing a prior art cooling fan structure;

Figure 2 is a front view of the cooling fan 70 shown in Fig. 1;

Figure 3 is an explanatory view showing the section of the fan blade taken along the line III-III of Fig. 2;

Figure 4 is a schematic sectional view 75 showing a cooling fan fitting structure according to a preferred embodiment of the present invention;

Figure 5 is a sectional view taken along the line V-V of Fig. 4;

80 Figure 6 is an explanatory view showing the characteristics of the fan blade shown in Fig. 4;

Figure 7 is an explanatory view showing 85 conventional relationships between the elevation angle and the lifting coefficient of a blade;

Figure 8 is a graph showing the relationships between the fan air flow rate and pitch angle of the fan blade on the basis of the 90 results of experiments;

Figure 9 is a graph showing the relationships between the pitch angle of the fan blade and the noise level on the basis of the results of experiments;

95 Figure 10 is a front view showing a cooling fan according to another embodiment of the present invention;

Figure 11 is a sectional view of the cooling 100 fan taken along the line VI-VI in Fig. 10;

Figure 12 is a side view showing a cooling fan according to still a further embodiment of the present invention; and

Figure 13 is a front view of the cooling fan 105 shown in Fig. 12.

Fig. 8 illustrates the relationships between the fan air flow rate and the pitch angle at the base portion of a fan blade which is effective as a rotational blade, in the case of a prior art axial flow type cooling fan for an automotive

110 vehicle according to the experiments of the present inventors. In the experiments, the rotational speed of the cooling fan was 3,000 r.p.m.

When the pitch angle θ of the fan blade is 115 set from 60° to 70° at the base portion thereof which is effective as a rotational blade, the fan air flow rate is maximum. When the pitch angle at the base portion of the fan blade is more than 70°, the mass rate of air 120 flow decreases. About 72° is the limit pitch angle for the purpose of introducing the effective air flow by the cooling fan.

Fig. 9 illustrates the relationships between the fan noise per unit air flow and the pitch 125 angle θ at the base portion of the fan blade.

When the pitch angle θ at the base portion of the fan blade is less than 12°, the fan noise is too much. As the pitch angle increases, the fan noise decreases, gradually. When the 130 pitch angle is set between 60° and 70°, the

fan noise becomes minimum. It is also discovered that the fan noise again increases when the pitch angle at the base portion of the fan blade is more than 70°.

5 The similar results were obtained in other experiments with fan rotational speeds of 2,000 and 3,000 r.p.m.

Figs. 4 to 6 show a preferred embodiment of the present invention. The cooling fan 3 includes the boss portion 5 and a plurality of circumferentially spaced fan blades 4 extending radially outwardly from the boss portion 4. The coupling 7, the fan shaft 6 and other parts may be of conventional construction.

10 The pitch angle of the fan blades 4 is set within a proper range in view of the foregoing test results. That is, the pitch angle of the fan blades 4 is so large at the base portion thereof which is effective as a rotational blade 20 that a cone-like air current can be produced, the imaginary fine point of which is positioned upstream of the cooling fan. For example, such an effective base portion of the fan blades 4 is within the distance r_1 from the 25 center of the cooling fan 3 where r_0 is the maximum radius of the boss portion 5 of the cooling fan 3 cooling fan 3 and r_1 is from 1.1 r_0 to 1.2 r_0 as shown in Fig. 4. For the best results, the pitch angle θ at the effective base 30 portion of the fan blades is set between 60° and 70°. The pitch angle becomes gradually small from the base portion of the fan blade to the head thereof. When the pitch angle of the fan blade is set in such a way, a cone-like 35 air current occurs so that the air flow not interfere with the engine and others so much, downstream of the cooling fan, as compared with the prior art axial air flow type cooling fan. Thus, the maximum air flow rate can be 40 obtained without fan noise.

In this connection, the circumferential velocity of the fan blade is low within the distance r_1 (1.1 r_0 to 1.2 r_0) from the center of the cooling fan to such a degree that the fan 45 noise and the fan air flow can be disregarded. For such a reason, the pitch angle θ is set substantially large at the pitch angle effective base portion 4a of the fan blade 4 except the portion within the distance r_1 from the center 50 of the fan. The pitch angle θ additionally becomes gradually smaller toward the head 4b of the fan blade 4.

Referring to Fig. 6, the air current will be taken into consideration. At the distance r 55 from the center of the cooling fan 3, v is the velocity in the direction of the air flow by the rotation of the cooling fan 3, and v' is the velocity in the direction perpendicular to the rotational plane of the cooling fan 3. If L is 60 the angle between v and v' , v is equal to $v' \cos L$. Accordingly, although the air flows in the direction perpendicular to the rotation plane of the fan 3 from upstream of the fan, the air current runs obliquely or divergently 65 from the surface of the fan blade 4 so that the

air flow rate increases by $1/\cos L$ as compared with the prior art axial flow type cooling fan. As a result, the velocity q' of the composite air flow is larger than q so that the air flow 70 mass rate can be increased.

Also, at the distance r from the center of the cooling fan, s and θ are the elevation angle and pitch angle on the sectional plane in parallel to the air flow v , respectively. The 75 actual elevation angle and pitch angle on the sectional plane in parallel to the actual air flow v' caused by the rotation of the cooling fan 3 is s' and θ' , respectively. In such a case, v' and θ' are larger than v and θ , respectively. 80 Consequently, s is larger than s' . That is, even if the pitch angle θ of the fan blade is set large, the air flows to become divergent downstream of the cooling fan as a cone-shape air current so that the actual elevation 85 angle s' of the fan blade 4 is small so as to prevent the air flow rate from decreasing downstream of the cooling fan.

Referring to Figs. 4, 10 and 11, it is preferable that air flow guide means 8 is 90 formed on the boss portion 5 of the cooling fan 3 in order to introduce forcibly the air flow to the rear of the boss portion 5 so that the air pressure can be prevented from dropping downstream of the cooling fan 3. In this 95 embodiment, the air flow guide means 8 is composed of spiral grooves 9 formed on the circumferential surface of the boss portion 5 from the front edge thereof to the back edge thereof. Each groove 9 is located between the 100 adjacent fan blades 4 fixed on the boss portion 5.

Figs. 12 and 13 show a further embodiment of the present invention. A plurality of small auxiliary fans 10 are fixed on the front 105 end of the boss portion 5 by means of small screws 14, as the air flow guide means 8. In this embodiment, it is preferable that the appearance of the boss portion 5 of the cooling fan 3 is formed as a cannon ball 110 continuously connected with the boss portion 11 for the auxiliary fans 10 in order to make the air flow smooth downstream of the auxiliary fans 10. The reference numeral 13 denotes the shroud for the auxiliary fans 10.

115 If the air flow guide means 8 is provided in such a way, the whole air current downstream of the cooling fan 3 becomes in a cone-like shape, the imaginary fine point of which is positioned in front of the cooling fan so that 120 air resistance to the engine and others can decrease as compared with the prior art axial flow type cooling fan, thereby to increase significantly the air flow mass rate downstream of the cooling fan 3. In addition, as 125 the turbulent phenomenon of air flow by the air resistance and the reverse involved air flow as stated above can be avoided, the fan noise can be minimized. In particular, the air flow is forcibly introduced along the boss portion 5 of 130 the cooling fan 3 to the thereof by the air flow

guide means 8 so that the air pressure can be prevented from dropping at the rear of the boss portion of the cooling fan 3. Otherwise, the air pressure is apt to drop as the result of 5 the air flow becoming divergent backwards from the cooling fan 3.

According to a radiator cooling fan for an automotive vehicle, the pitch angle θ of the fan blade is set large at the base portion of 10 the fan blade which is effective as a rotational blade to such a degree that the air can flow in a cone shape, the imaginary fine point of which is positioned in front of the cooling fan. Therefore, the air resistance to the engine 15 and others located downstream of the cooling fan can be significantly decreased. In addition, the air flow guide means can be provided to prevent the air pressure from dropping at the rear of the boss portion of the 20 cooling fan. Thus, the air flow rate can be significantly increased. As a result, the cooling effects of the radiator can be improved. Assuming that the air flow mass rate is constant, the rotational speed of the cooling fan can be 25 decreased as compared with the prior art axial flow type cooling fan so that the fan noise can be minimized.

CLAIMS

30 1. A radiator cooling fan for an automotive vehicle, comprising:
 a boss portion rotatably provided at the center of the cooling fan; and
 a plurality of circumferentially spaced fan 35 blades extending radially outwardly from said boss portion, the pitch angle of said fan blades being so large at the base portion thereof which is effective as a rotational blade, as to produce substantially a cone-like air 40 current, the imaginary fine point of which is positioned upstream of the cooling fan, and the pitch angle thereof becoming gradually smaller from said base portion of said fan blades to the head thereof.

45 2. A radiator cooling fan for an automotive vehicle as claimed in claim 1 wherein the pitch angle of said fan blades is set between 60° and 70° at said base portion thereof.

3. A radiator cooling fan for an automotive vehicle as claimed in claim 1 or 2 further comprising air flow guide means formed on said fan boss portion for introducing the air flow backwards from said fan boss portion.

4. A radiator cooling fan for an automotive vehicle as claimed in claim 3 wherein said air flow guide means is a spiral groove formed on the circumferential surface of said boss portion.

5. A radiator cooling fan for an automotive vehicle as claimed in claim 3 wherein said air flow guide means is a small auxiliary fan provided at the front end portion of said fan boss portion.

6. A radiator cooling fan for an automotive vehicle, substantially as described with

reference to, and as illustrated in Figs. 4 and 5, or Figs. 10 and 11, or Figs. 12 and 13 of the accompanying drawings.

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